MECHANICAL ENGINEERING NEWS

COADE® Engineering Software

For the Power, Petrochemical and Related Industries

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BULLETIN POLICY

The COADE Mechanical Engineering News Bulletin is published on a quarterly basis from the COADE offices in Houston, Texas. The Bulletin is intended to provide information about software applications and development for Mechanical Engineers serving the power, petrochemical and related industries. Additionally the Bulletin will serve as the official notification vehicle for software errors discovered in those Mechanical Programs offered by COADE.

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PC VIRUSES

Until very recently, we did not consider PC viruses to be a major threat to the serious engineering user. By not using modems or networks, and by not installing "free" or "game" software, the PC is pretty much isolated. That's what we thought anyway. Viruses can unfortunately be a very big threat to the continued use of a PC and should be guarded against very closely. Guarding against virus infection is fairly straightforward providing a few simple safeguards are followed.

Viruses are not a problem on a PC until they become active. The mere presence of a virus alone is synonymous with the discovery of a tumor. The next question is whether or not the tumor is benign or malignant. Most viruses are benign. Unfortunately, the unaware user can convert a benign virus into a very malignant virus with only a few simple, common steps.

A virus is a part of a program that is in some way destructive. Since the virus is a program it must be loaded and executed before it can do any harm. A virus that simply resides benignly on the PC and is never executed, cannot do any damage. It is

We are also looking at providing a 10CFR50 Appendix B option to users that require further documentation of errors and omissions. A full QA document for **CAESAR II** is also being prepared that can be purchased.

SOFTWARE SUPPORT AND CONSULTING

At COADE we are frequently asked to review designs, model input and various types of analyses results. There are two groups at COADE that provide different levels of this service. The first group provides support for COADE's software products. This group is comprised of mechanical engineers that have heavy analysis, software development, and training backgrounds. This groups' primary goal is to make sure that the COADE software is being properly utilized. The budget for this group comes from software maintenance and updates.

The second group provides various consulting and design services for COADE clients. The budget for this group comes from consultation fees paid by clients on a job-by-job basis. This group is comprised of mechanical engineers that have heavy analysis, design and project oriented backgrounds.

As an example: the software support group will often review a client's dynamic input to make sure that nothing unusual has been entered, i.e. to make sure that an impulse load problem is not being solved with the harmonic solver, etc. Reviewing a faxed copy of a typical dynamic input will not take longer than about 5 minutes. The consulting support group will often review the same input and check everything, i.e. make sure that the correct equations for dynamic loads are used, and that the correct value has been computed and entered properly; that the model is sufficiently detailed to observe the dynamic response for the type of loading experienced, etc.

The consulting support group handles the full range of piping and pressure vessel design from drafting thru installation, and trouble-shooting.

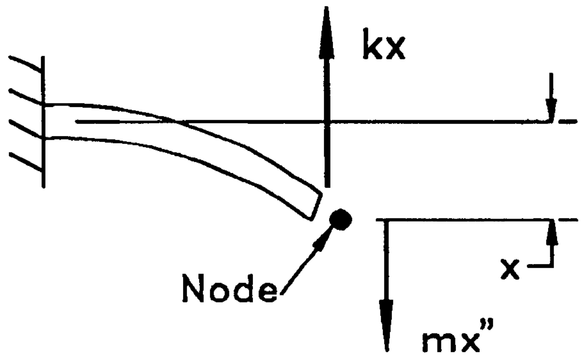
It is a common misconception that COADE is a group of computer programmers. COADE employees 11 full-time Mechanical Engineers with over 140 years of experience in the piping and pressure vessel industries.

CAESAR II DYNAMIC QUESTIONS AND ANSWERS

Q: Why does the **CAESAR II** output not reflect the mx" term, where (m) is the lumped mass at the node and (x") is the maximum acceleration at the node?

ANS: A look at a sketch of the forces acting on a node in a dynamic problem helps explain why the mass-acceleration product should not be included in the element/support load calculations.

CANTILEVER GEOMETRY



k – Stiffness of Beam
x – Vertical Displacement
m – 1/2 of Mass of Cantilever
mx" – Acceleration of Vertical Direction

FIGURE 1

The mx" and the kx terms represent equal and opposite forces that act on the node at the end of the element. The total force on the element producing stress throughout the element is kx <or> mx". The force at the end of the element should be kx, not (kx) + (mx").

Q: But shouldn't the inertia effects (mx") be considered in order to accurately describe the internal loadings and to satisfy dynamic equilibrium.

ANS: The inertial effects ARE included. It is the inertial effects that are offset by the elastic effects. The equation of motion "in words" should clarify this:

Inertial Effects + Elastic Effects + Dynamic Forces = 0.0

Q: If the mass is included correctly why does adding more elements along the run, i.e. "distributing" the mass, result in smaller internal forces and moments in the system?

ANS: Breaking single elements down into a larger number of small elements improves the mass model of the piping system, not how the inertial effects are handled. If the breaking down of the straight runs in a piping system results in different calculated values, (primarily the stresses and maximum displacements), then the cruder mass model was not sufficient to include all of the modal effects that are associated with the higher modes of vibration.

Q: Is the handling of the inertial effect the same as the missing mass correction calculation?

ANS: No. The missing mass calculation does not really include the effects of a poorly mass modeled piping system either. Missing mass effects reflect inertial loads on parts of the piping system that are not "dynamically" excited because of their rigidity, but that are inertially loaded because of their attachment to a point on the structural system that is "dynamically excited".

Q: In statics, when there are two elements attached to a single node, the forces and moments acting on the elements on either side of the node are equal and opposite unless there is a restraint at the node. Why in dynamics is this not true?

ANS: The dynamic force equation where two elements frame into the same node and there is no externally applied dynamic forcing function at the node is shown below:

$$(K1)(x) + Mx'' + (K2)(x) = 0.0$$

Where: K1 is the elastic stiffness due to element 1.

K2 is the elastic stiffness due to element 2.

x is the maximum displacement at the common node between the two elements.

M is the mass from each of the elements that is lumped at the common node.

x" is the maximum acceleration at the common node.

The difference in the dynamic force is the mass times maximum-acceleration product for the common node. Note that the stresses at the common node on element 1 are due to the externally acting force (K1)(x), and the stresses at the common node on element 2 are due to the externally acting force (K2)(x).

DISCUSSION:

In many cases, questions about dynamic analyses can be answered by the running of small example jobs. This approach illustrates the numerical and practical characteristics of dynamic analysis, and the **CAESAR II** method for inputting and observing these characteristics. Often times confusing dynamic results seem very clear when the system exhibiting the results is simple. (Unfortunately many **CAESAR II** users start their dynamics training on a "live", large, very complex static model, and become overwhelmed by the abundance of dynamic results that are printed.)

Every dynamic result should "make sense" to the user, and in turn every dynamic result should be "looked at", and the "make sense" test applied. Many users look only at dynamic displacements and stresses, ignoring participation factors, mode reports, nonlinear boundary condition reports, and mass distribution reports. Many errors in a dynamic analysis can be uncovered by reviewing each of these report and being sure that the values reported "make sense" as far as the system and its loads are concerned.

IF A PARTICULAR NUMBER IN A CAESAR II DYNAMIC REPORT DOES NOT MAKE SENSE, THEN THERE IS EITHER A MISUNDERSTANDING OF THE VALUE PRINTED, AN ERROR IN THE INPUT, OR AN ERROR IN **CAESAR II**. In any case, the source of the confusion should be identified and explained.

COADE provides two levels of dynamic review for users of **CAESAR II**. Typically via fax, a software analyst can spend a short time, (usually no more than 15 minutes) looking over the dynamic input, pointing out obvious errors or omissions. Very little time is needed for this "review" because the dynamic input to **CAESAR II** does not require much data entry. The second level of dynamic review involves a thorough check of the modeling and dynamic assumptions made in the model setup and dynamic analysis. For this review, complete drawings and **CAESAR II** input are required. This check is done by the COADE consulting engineering group, and involves the payment of a consulting fee. Similarly, the consulting engineering group can analyze dynamic problems from start-to-finish if the user so desires.

Users of the water-hammer, slug flow, and relief load dynamic analysis tools in **CAESAR II**, can be content with the knowledge that the method has now been used extensively throughout industry with very satisfying results. Slug flow systems have been designed, snubber failures that did occur have been predicted, relief-safety system designs have been verified by live tests, observed displacements have been repeatedly predicted by **CAESAR II** simulations, dynamic sensitivities to periodic excitation have been modeled and the sensitivity eliminated by the mathematical installation of supports rather than by a trial-and-error field solution, and measured displacements due to wind loads have been modeled using the **CAESAR II** dynamic wind gust load analysis. Users often ask how good the frequency domain solutions for water-hammer, slug flow and relief loads are. The answer is that whenever the predominant excitation is of an impulse type, the solutions are very good. Fortunately, the excitation mechanism for the problems described above is impulsive.

DOUBLE ROD MODELING

The figure below shows a situation where the double rods provide both "Y" and "RZ" support. In most cases this type of support condition is modeled with only a "Y" restraint at the pipe center line. The "RZ" restraint is ignored. For stiffer systems, i.e. short configurations or large diameter lines where there is a tendency for the system to have "Z" rotations, this can result in an errant redistribution of the "Z" moments in the piping model.

The first modeling improvement might appear as shown in Figure 4.

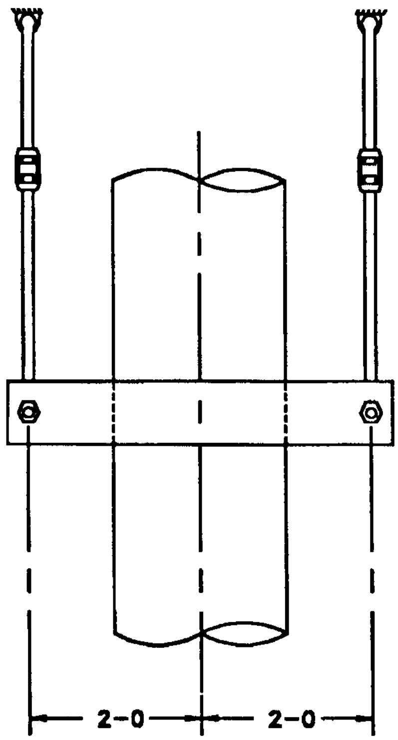


FIGURE 2

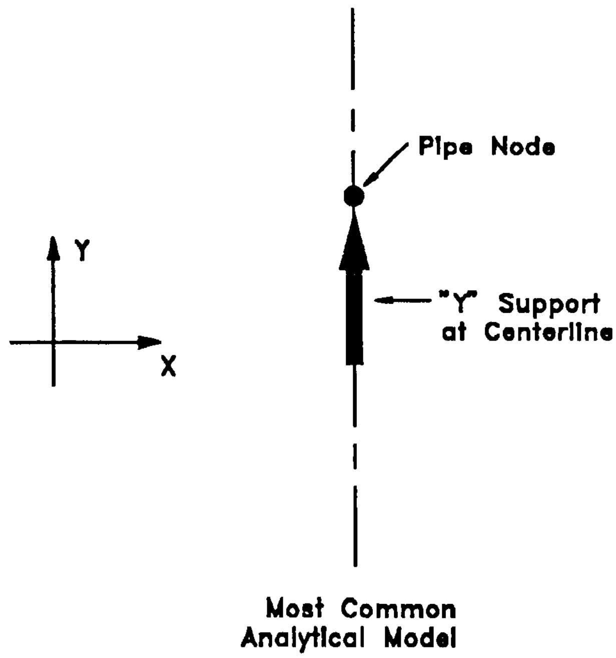


FIGURE 3

Unfortunately when there is a tendency for a stiff system to rotate about the "Z" axis this model can produce errant results also. Quite often one of the rods will be carrying a small compressive load or no load at all! This is certainly not a desirable design condition. Improper computer modeling, or evaluation of the results, does not reveal the tendency for this support to redistribute its loads so unevenly.

When this condition is discovered at the design phase the support should either be moved to a more favorable location, or redesigned such that there is no significant "RZ" restraint provided. Several items need to be considered if the analyst needs to model an existing support of this type where there is the tendency to rotate in-plane.

1) Are there also large, horizontal translations? If so then large rotation, rod type, restraints must be used.

2) When the model shown in Figure 4 is built, is one of the supports on either side of the pipe in compression? If not, then a more sophisticated model is not necessary.

3) In the Figure 4 model, was the actual axial stiffness of the rods (AE/L) used for the restraint stiffness? If not it should be. Very stiff models react very well to situations near small drops in stiffness.

4) How much "play" is there in the supporting rod mechanism? Before one of the rods goes into compression, this "play" is taken out of the system. Sometimes, a very little gap can eliminate a large compressive load in a rod due to thermally induced rotations. Adding this gap to the analytical model of an "as installed" system can result in a large reduction in the load carried by the supports, and a large reduction in the induced moment on the pipe at the support.

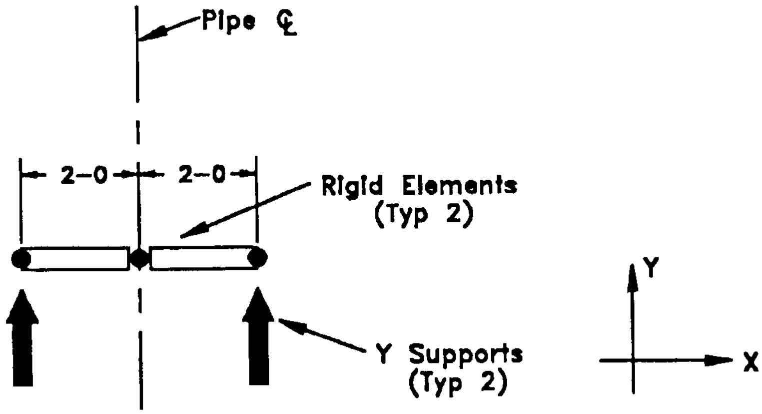
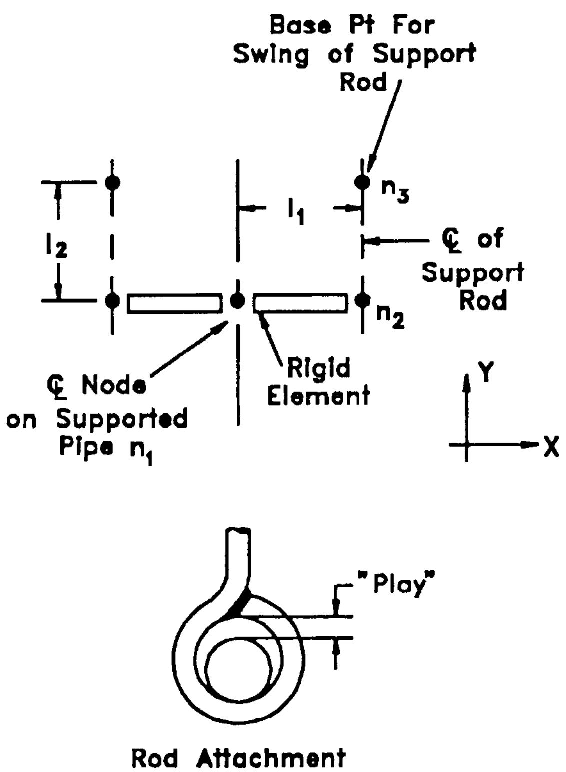


FIGURE 4

The model shown in the figure below is the most comprehensive **CAESAR II** model typically needed for this assembly. (Note, the sample coding below only shows the right rod support. Similar entries are required for the left rod.)



Sample Support Nodes

Rigid Element	From	n_1	Rigid Y Weight = 0	
	To	n_2		
	DX	l_1		
Rod Restraint	Restraint Y			
	Node	n_2	Cnode	n_3
	Type	YROD		
	Stif	AE/L		
	Len	l_2		
Horizontal Stops Θ Pivot	Node	n_3		
	Type	X		
	Node	n_3		
	Type	Z		
Allows Possible "Play" In The Rods	Node	n_3		
	Type	+Y		
	Node	n_3		
	Type	−Y		
	Gap	"Play"		

FIGURE 5